

# JOURNAL OF MECHANICAL ENGINEERING

*An International Journal*

VOLUME 12 NO. 2

DECEMBER 2015 / ISSN 1823-5514



# JOURNAL OF MECHANICAL ENGINEERING

*An International Journal*

## EDITOR IN CHIEF:

Professor Wahyu Kuntjoro – Universiti Teknologi  
MARAs, Malaysia

## EDITORIAL BOARD:

Datuk Professor Ow Chee Sheng – Universiti  
Teknologi MARA, Malaysia

Dr. Ahmad Azlan Mat Isa – Universiti Teknologi  
MARAs, Malaysia

Dr. Faqir Gul – Institute Technology Brunei,  
Brunei Darussalam

Dr. Mohd. Afian Omar – SIRIM Malaysia

Dr. Valliyappan David a/l Natarajan – Universiti  
Teknologi MARA, Malaysia

Dr. Yongki Go Tiauw Hiong – Florida Institute  
of Technology, USA

Professor Abdelmagid Salem Hamouda – Qatar  
University, Qatar

Professor Abdul Rahman Omar – Universiti  
Teknologi MARA, Malaysia

Professor Ahmed Jaffar – Universiti Teknologi  
MARAs, Malaysia

Professor Bernd Schwarze – University of  
Applied Science, Osnabrueck, Germany

Professor Bodo Heimann – Leibniz University  
of Hannover Germany

Professor Darius Gnanaraj Solomon – Karunya  
University, India

Professor Hazizan Md. Akil – Universiti  
Sains Malaysia, Malaysia

Professor Mohd. Zulkifly Abdullah –  
Universiti Sains Malaysia, Malaysia

Professor Roslan Abd. Rahman – Universiti  
Teknologi Malaysia, Malaysia

Professor Salmiah Kasolang – Universiti  
Teknologi MARA, Malaysia

Professor Essam E. Khalil – University of Cairo,  
Egypt

Professor Ihsan S. Putra – Bandung Institute of  
Technology, Indonesia

Professor Shahrir Abdullah – Universiti  
Kebangsaan Malaysia

Professor Shahrum Abdullah – Universiti  
Kebangsaan Malaysia

Professor Masahiro Ohka – Nagoya University,  
Japan

Professor Mirosław L Wyszynski – University of  
Birmingham, UK

Professor Mohamad Nor Berhan – Universiti  
Teknologi MARA, Malaysia

Professor P. N. Rao – University of Northern  
Iowa, USA

Professor Wirachman Wisnoe – Universiti  
Teknologi MARA, Malaysia

Professor Yongtae Do – Daegu University, Korea

## EDITORIAL EXECUTIVES:

Dr. Solehuddin Shuib

Dr. Muhad Rozi Mat Nawi

Dr. Noor Azlina Mohd Salleh

Dr. Siti Mariam Abdul Rahman

Nurul Hayati Abdul Halim

Rosnadiyah Bahsan

Copyright © 2015 by the Faculty of Mechanical Engineering (FKM), Universiti Teknologi MARA, 40450 Shah Alam, Selangor, Malaysia.

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted in any form or any means, electronic, mechanical, photocopying, recording or otherwise, without prior permission, in writing, from the publisher.

*Journal of Mechanical Engineering (ISSN 1823-5514) is published by the Faculty of Mechanical Engineering (FKM) and UiTM Press, Universiti Teknologi MARA, 40450 Shah Alam, Selangor, Malaysia.*

*The views, opinions and technical recommendations expressed herein are those of individual researchers and authors and do not necessarily reflect the views of the Faculty or the University.*

# JOURNAL OF MECHANICAL ENGINEERING

*An International Journal*

|               |               |                |
|---------------|---------------|----------------|
| Vol. 12 No. 2 | December 2015 | ISSN 1823-5514 |
|---------------|---------------|----------------|

1. The Design of a High Aspect Ratio HALE Aircraft Composite Wing. Part I: Static Strength Analysis 1  
*Bambang K. Hadi*  
*Muhammad A. Ghofur*  
*IndraPermana*
2. The Design of a High Aspect Ratio HALE Aircraft Composite Wing. Part II: Buckling and Flutter Speed Analysis 13  
*Bambang K. Hadi*  
*IndraPermana*
3. Analysis on the Factors and Effects of Engine Tapping Noise by Design of Experiment (DOE) 27  
*Mohd Shahrudin Mohd Alias*  
*Nor Hayati Saad*  
*Juri Saedon*  
*Nor Hidayah Hassan*  
*Saidi Harun*
4. Computer Aided Engineering Approach to the Development of a Lawnmower Blade: A Reverse Engineering Application to Reduce Noise Levels 49  
*Trishel Gokool*  
*Fahraz Ali*  
*Boppana V. Chowdary*  
*Kimberly Kanchan*

5. Low Energy Impact on the Short Kenaf Fibre Reinforced Epoxy Composites: Effect to the Residual Strength and Modulus 71  
*Syarifah Yunus*  
*Abdul Hakim Abdullah*  
*Nurul Hayati Abdul Halim*  
*Zuraidah Salleh*  
*Yakub Md. Taib*
6. Numerical Investigation of Heat Transfer and Fluid Flow Characteristics of an Internally Ribbed Tube 85  
*Kailash Mohapatra*  
*Dipti Prasad Mishra*



# Numerical Investigation of Heat Transfer and Fluid Flow Characteristics of an Internally Ribbed Tube

*Kailash Mohapatra*

*Department of Mechanical Engineering,  
Raajdhani Engineering College, Bhubaneswar, India 751 017  
kailash72@ymail.com*

*Dipti Prasad Mishra\**

*Department of Mechanical Engineering,  
Birla Institute of Technology Mesra, Ranchi, India 835 215*

*\*Corresponding Author  
dpmishra@bitmesra.ac.in*

## ABSTRACT

*Numerical investigation have been performed by solving the conservation equations of mass, momentum, energy with two equation based k-e model to determine the wall temperature, outlet temperature and Nusselt number of an internally ribbed tube. It has been found from the numerical investigation that the heat transfer and pressure drop is increased by increasing the rib height and rib number. It was also found that there exists an optimum angle of inclination for maximum heat transfer and minimum pressure drop. The heat transfer is found to be highest in triangular rib compare to other shape.*

**Keywords:** *ribbed tube, Nusselt number, pressure drop, optimum*

## Introduction

Internal ribs are extensively used in internal surfaces of heat exchangers tube to improve the heat transfer rate. These ribs break the laminar sub-viscous boundary layer formed on the heated surface and create artificial turbulence around the ribs region. Due to this artificial turbulence the heat transfer to the fluid is increased and the wall temperature falls significantly. The heat transfer and fluid flow characteristics of internal ribs significantly depends on ribs geometry and ribs number.

**Nomenclature:**

|                         |  |
|-------------------------|--|
| $C_{1e}$ :              | constant value of 1.44   |
| $C_{2e}$ :              | constant value of 1.92   |
| D:                      | diameter of tube   |
| f:                      | friction factor  |
| H:                      | height of the rib  |
| h:                      | heat transfer coefficient  |
| k:                      | turbulent kinetic energy   |
| L:                      | length of the tube   |
| n:                      | number of fins   |
| Nu:                     | Nusselt number   |
| p:                      | pressure   |
| P:                      | perimeter  |
| $Pr_t$ :                | turbulent Prandtl number   |
| q:                      | heat flux  |
| Re:                     | Reynolds number  |
| $S_{ij}$ :              | $\frac{1}{2} \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right)$ |
| T:                      | temperature  |
| $T_{sx}$ :              | Axial wall temperature   |
| t:                      | time   |
| U:                      | velocity   |
| $-\overline{u_i u_j}$ : | $2 \frac{\mu_t}{\rho} S_{ij}$  |
| w:                      | width of the rib   |
| x:                      | axial location   |

**Greek symbols**

|                        |  |
|------------------------|--|
| $\alpha$ :             | thermal diffusivity                            |
| $\varepsilon$ :        | rate of dissipation                            |
| $\mu$ :                | dynamic viscosity                              |
| $\mu_t$ :              | turbulent viscosity                            |
| $\nu$ :                | kinematic viscosity                            |
| $\phi$ :               | scalar variable<br>either $k$ or $\varepsilon$ |
| $\rho$ :               | density  |
| $\sigma_k$ :           | turbulent Prandtl number<br>for $k$            |
| $\sigma_\varepsilon$ : | turbulent Prandtl number<br>for $\varepsilon$  |

**Subscripts**

|            |                        |
|------------|------------------------|
| avg:       | average                |
| B:         | bulk                   |
| $\infty$ : | ambient                |
| sx:        | axial surface distance |
| x:         | location               |

Considerable research works have been conducted both numerically and experimentally to study the heat transfer and fluid flow characteristics of different types of duct having internal ribs. Webb et al. [1] developed correlations for heat transfer and friction factor for turbulent flow inside the tube having repeated ribs which are based on heat-momentum transfer analogy to a flow over a rough surface. Those correlations were compared with experimental data and it was found that those correlations were superior to other methods, especially in accounting for the effect of Prandtl number. A Particle Image Velocimetry (PIV) measurement have been conducted Wang et al. [2] to study the performance of a channel with periodic ribs on one wall for a Reynolds number 22000 based on the bulk mean velocity and channel height. It was found from the study that the maximum Reynolds shear stress occurred at the leading edge of the rib because of ejection motions. Numerical investigation performed by Hossainpour and Hassanzadeh [3] to determine the heat transfer and fluid flow characteristics for a turbulent flow inside a tube having helical ribs. They concluded that by providing helical ribs there is a significant effect on heat transfer augmentation and pressure drop. Numerical and experimental investigations were conducted by Peng et al. [4] to determine the convection heat transfer in a channel with  $90^\circ$  ribs and V-shaped ribs. They concluded that the overall thermal and hydraulic performance of V-shape ribs was better than  $90^\circ$  ribs. Experimental investigation have been performed by Wen-Tao et al. [5] to determine heat transfer and friction factor in an internally grooved tube for a turbulent flow condition. They have developed correlations to determine the heat transfer which is very helpful for engineering design. Yadav and Bhagoria [6] conducted numerical investigation for turbulent flow to study the effect of circular ribs placed at the inner side of a duct with different height. They concluded that the thermo-hydraulic performance was best at a particular rib pitch to rib height ( $P/e = 10.71$ ). Yongsiri et al. [7] concluded from the numerical investigation, inclined detached ribs with inclination angle  $60^\circ$  and  $120^\circ$  had higher heat transfer and thermal performance compared to other inclination angle for high Reynolds number flow. Experimental investigation performed by Shivkumar et al. [8] for a divergent rectangular duct having different sized square ribs to study thermal and fluid flow characteristics. It was found that rib with 3 mm height had highest heat transfer rate compared to 6 mm and 9 mm height ribs and the pressure drop with 6 mm and 9 mm rib height was more than with 3 mm height rib. Experimental investigation also shows for a rectangular duct the thermal performance is better in case of wedge shaped ribs and V-shaped ribs compare to a smooth duct (Aharwal et al. [9], Pankaj et al. [10]).

From the earlier works cited in the literature, there is a clear evidence that the thermal performance is improved by providing internal ribs inside the duct. In the present work the heat transfer and fluid flow characteristics of an internal ribbed tube have been numerically analyzed for a flow which is assumed to be turbulent and is developing in nature. The mass flow rate inside the ribbed tube

is maintained 0.0196 kg/s and the nominal diameter of the tube is 0.07 m. So the corresponding Reynolds number based on the nominal diameter is 20000 and the entrance length was found to be 1.6 m. In the present investigation the tube length is considered to be 1 m, which indicates the flow is not fully developed rather it is developing in nature and it is only developed by 65%. The ribs have been attached at the inside wall of a tube and the heat transfer and fluid flow characteristics have been investigated by varying the rib number, shape and size. All the computations have been performed using Fluent 14 for different flow conditions to determine the maximum heat transfer.

## Mathematical Formulation

The computational investigation is carried out for an internally finned tube of diameter,  $D$  and length,  $L$  as shown in Figure 1a. Air is allowed to enter into the tube from one end at room temperature where as the other end is exposed to the surrounding atmosphere. Ribs are attached at the inner wall of the tube. The investigation is initiated with the semi-circular ribs of radius  $r$  as shown in Figure 1a and subsequently the rib number, shape and sizes have been changed while the mass flow rate of air inside the tube was kept constant. The flow field in the domain would be computed by using incompressible, 2D axi-symmetric, Navier-Stokes equations with a two equation based k- $\epsilon$  turbulence model along with the energy equations. The fluid used in the simulation is air, at temperatures of 300 to 500 K and the flow is treated to be incompressible.

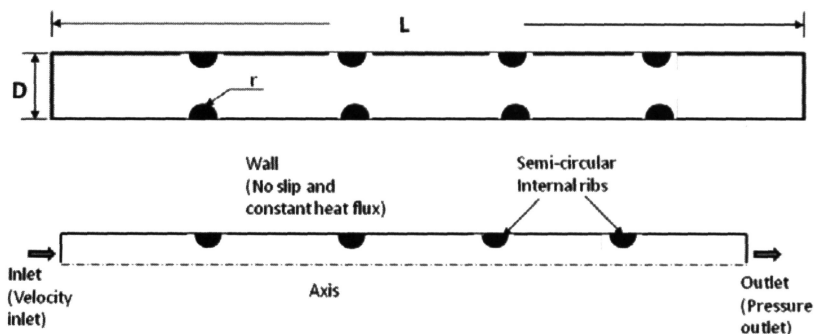


Figure 1: (a) Tube with rectangular internal ribs and (b) Computational domain of tube with internal ribs and the boundary conditions applied to it

## Governing equations

The governing equations for the above analysis can be written as:

### Continuity

$$\frac{\partial}{\partial x_i}(\rho U_i) = 0 \quad (1)$$

### Momentum

$$\frac{D(\rho U_i)}{Dt} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \rho \overline{u_i u_j} \right] \quad (2)$$

$$\frac{p}{\rho} = RT \quad (2a)$$

Variable R is the characteristic gas constant and is equal to 0.287 kJ/kg-K for air.

The density  $\rho$  is taken to be a function of temperature according to ideal gas law as per eqn. (2a), while dynamic viscosity  $\mu$  is kept constant and  $-\overline{u_i u_j}$  is expressed as:

$$-\overline{u_i u_j} = 2 \frac{\mu_t}{\rho} S_{ij}, \quad S_{ij} = \frac{1}{2} \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \quad (3)$$

### Energy Equation

$$\frac{D(\rho T)}{Dt} = \frac{\partial}{\partial x_i} \left[ \left( \frac{\mu}{Pr} + \frac{\mu_t}{Pr_t} \right) \frac{\partial T}{\partial x_i} \right] \quad (4)$$

### Conduction Equation

$$\frac{\partial}{\partial x_i} \left( k \frac{\partial T}{\partial x_i} \right) = 0 \quad (5)$$

**Turbulence Kinetic Energy - k**

$$U_j \frac{\partial k}{\partial x_j} = \frac{\mu_t}{\rho} S^2 - \varepsilon + \frac{\partial}{\partial x_j} \left[ \frac{1}{\rho} \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] \quad (6)$$

Where S is the modulus of the mean rate-of-strain tensor, defined as

$$S \equiv \sqrt{2S_{ij}S_{ij}} \quad (7)$$

**Rate of Dissipation of k**

$$U_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\varepsilon}{k} \left( C_{1\varepsilon} \frac{\mu_t}{\rho} S^2 - C_{2\varepsilon} \varepsilon \right) + \frac{\partial}{\partial x_j} \left[ \frac{1}{\rho} \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] \quad (8)$$

$\mu_t$  is expressed as

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (9)$$

$\sigma_k$  and  $\sigma_\varepsilon$  are the Prandtl numbers for  $k$  and  $\varepsilon$ .

The constants used in the above k-ε equations are the following:

$$C_{1\varepsilon} = 1.44; C_{2\varepsilon} = 1.92; C_\mu = 0.09; \sigma_k = 1.0; \sigma_\varepsilon = 1.3, \text{Pr}_t = 1$$

**Boundary Conditions**

The boundary conditions for ribbed tube have been shown in Figure 1b. The tube walls are solid and have been given a no-slip and constant heat flux boundary condition. Pressure outlet boundary condition has been imposed at the outlet of the tube where as velocity inlet boundary condition has been employed at the entrance where the air was allowed to enter into the tube at room temperature. Ribs are solid and fixed at the inner wall of the tube, so no-slip and coupled boundary condition has been applied to all the surfaces of the ribs lying inside the tube. As the ribs are placed symmetrically at the inner surface of the tube so, axi-symmetric boundary condition has been used.

At the pressure outlet boundary, the velocity will be computed from the local pressure field so as to satisfy continuity but all other scalar variables such as,  $T$ ,  $k$  and  $\epsilon$  are computed from the zero gradient condition, Dash [11].

The turbulent quantities,  $k$  and  $\epsilon$ , on the first near wall cell have been set from the equilibrium log law wall function as has been described by Jha and Dash [12, 13] and Jha et al. [14]. Generally the  $k$ - $\epsilon$  turbulent model is performed well for confined flows where Reynolds shear stresses are most important (Versteeg and Malalasekera, [15]). The turbulent intensity at the inlet of the tube has been set to 2%, with the inlet velocity being known, and the back flow turbulent intensity at all the pressure outlet boundary have been set to 5%. If there is no back flow at a pressure outlet boundary then the values of  $k$  and  $\epsilon$  are computed from the zero gradient condition at that location.

### Computations of some important heat transfer and fluid flow parameters

The heat transfer and fluid flow parameters are computed as given in Islam and Mozumder [16] and Yu et al. [17].

Local Nusselt number for the ribbed heated tube.

$$Nu_x = \frac{h_x D}{k_{air}} \quad (10)$$

Local heat transfer coefficient has been computed as.

$$h_x = \frac{q}{T_{sx} - T_{Bx}} \quad (11)$$

Where  $h_x$ ,  $D$ ,  $k_{air}$ ,  $T_{sx}$  and  $T_{Bx}$  are respectively, the local heat transfer coefficient, internal nominal diameter of tube, thermal conductivity of air, local tube wall temperature and local bulk temperature of fluid.

$$T_{Bx} = T_i + \frac{q P_n}{\dot{m} C_p} x \quad (12)$$

$T_i$ ,  $q$ ,  $P_n$ ,  $x$ ,  $C_p$  and  $\dot{m}$  are respectively, inlet fluid temperature, wall heat flux, nominal perimeter, axial position, specific heat and mass flow rate of fluid.

$$\dot{m} = \rho V_{in} A \quad (13)$$

$\rho$ ,  $V_{in}$  and  $A$  are the density, inlet fluid velocity and area of tube at the inlet respectively.

The Reynolds number has been computed based on nominal diameter of the tube based on inlet flow temperature and is given by

$$Re = \frac{V_{in} D}{\nu_{air}} = \frac{4m}{\mu P} \quad (14)$$

$\nu_{air}$  is the kinematic viscosity of air based on inlet condition

$$Nu_{avg} = \frac{h_{avg} D}{k_{air}} \quad (15)$$

$$h_{avg} = \frac{q}{T_{savg} - T_{Bavg}} \quad (16)$$

$$T_{Bavg} = \frac{\sum_{j=1}^n T_{Bx_j}}{n} \quad (17)$$

$$T_{savg} = \frac{\sum_{j=1}^n T_{sx_j}}{n} \quad (18)$$

Where  $Nu_{avg}$ ,  $h_{avg}$ ,  $T_{savg}$  and  $T_{Bavg}$  are respectively, the average Nusselt number, average heat transfer coefficient, average wall temperature and average bulk temperature.

$$f_x = 2\Delta P \frac{D}{x} \frac{1}{\rho V_{in}^2} \quad (19)$$

Where  $f_x$ ,  $\Delta P$ ,  $r$  and  $x$  are local friction factor, static pressure difference from entrance to a distance  $x$  in the downstream from the entrance, density and axial distance along the tube respectively.



## Numerical Solution Procedure

Steady two-dimensional axi-symmetric equations of mass, momentum, energy and turbulence have been solved by the algebraic multi grid solver of **Fluent 14** in an iterative manner by imposing proper boundary conditions. A first order upwind scheme (for convective variables) was considered for momentum as well as for the turbulent discretized equations. After a first-hand converged solution could be obtained the scheme was changed over to second order upwind scheme. However, before switching to the higher order scheme the meshes near the tube wall and near the fin surface area were refined to half of its original size. As the density is taken as a function of temperature according to ideal gas equation therefore SIMPLE algorithm with PRESTO (Pressure Staggered Option) scheme has been used for better convergence. Under relaxation factors (0.3 for pressure, 0.7 for momentum and 0.8 for  $k$  and  $\epsilon$ ) were used for the convergence of all the variables. Convergence of the discretized equations were said to have been achieved when the whole field residual for all the variables fell below  $10^{-3}$  except energy equation and for energy equation residual was set  $10^{-6}$ .

## Results and Discussion

### Grid sensitive test:

Grid sensitivity test has been shown in Figure 2 for a sample case of internal ribbed tube. The tube length and diameter were considered (with reference to Islam and Mozumder [16]) to be 1 m and 0.07 m respectively and the wall of the tube is subjected to a constant heat flux of  $3000 \text{ W/m}^2$ . Five numbers of semi-circular ribs of radius 10 mm have been attached to the internal wall of the tube. Initially coarse grids have been used and subsequently cells were refined and it was seen that after refining the entire meshes inside the tube from 500 to 9000, the outlet temperature was decreased by 1%. After that the cells near the walls of the tube and ribs were refined (a cutaway view of the tube can be seen from Figure 3) so that the mesh number increased to 15000. It was found that after increasing the cells more than 9000 the outlet temperature was not changed significantly. As the finer grids do not improve the outlet temperature, one can use 9000 cells (little larger grid corresponding to 9000 number of cells) for better results. When the shape and size of the internal fins were changed the grid test was performed before reporting any results.

temperature is dropped more and more. This suggests the heat transfer to air is increased with increase in the height of the rib. The case is more prominent in Figure 5, where the outlet temperature is increased with the height of the rib and more heat is transferred to air. It is also observed from the Figure 5 the outlet temperature has no significantly increment for  $r/D$  ratio above 0.225. It can be seen from the Figure 4 the wall temperature rise sharply towards the end of the tube for  $r/D$  ratio 0.28 and 0.35; this suggests the heat transfer from the heated wall is progressively reduced towards end of the tube due to the growth of boundary layer. As a result the exit temperature of the tube does not improve significantly for  $r/D$  beyond 0.225.

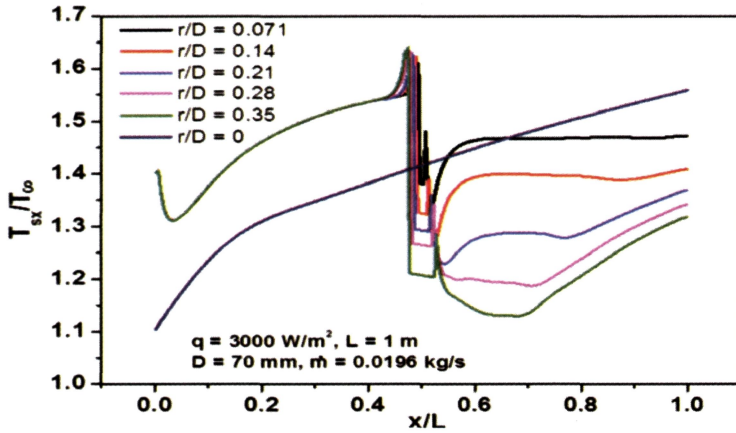


Figure 4: Wall temperature as a function of radius of semicircular rib height middle half of the tube. It can be seen from the plot the wall temperature

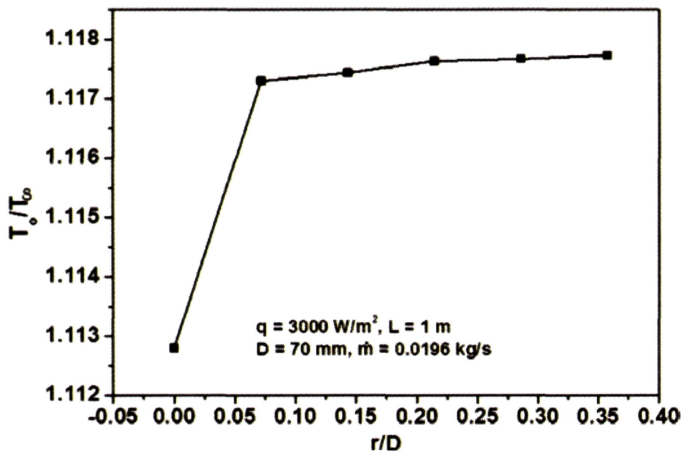


Figure 5: Outlet temperature as a function of radius of semicircular ribs

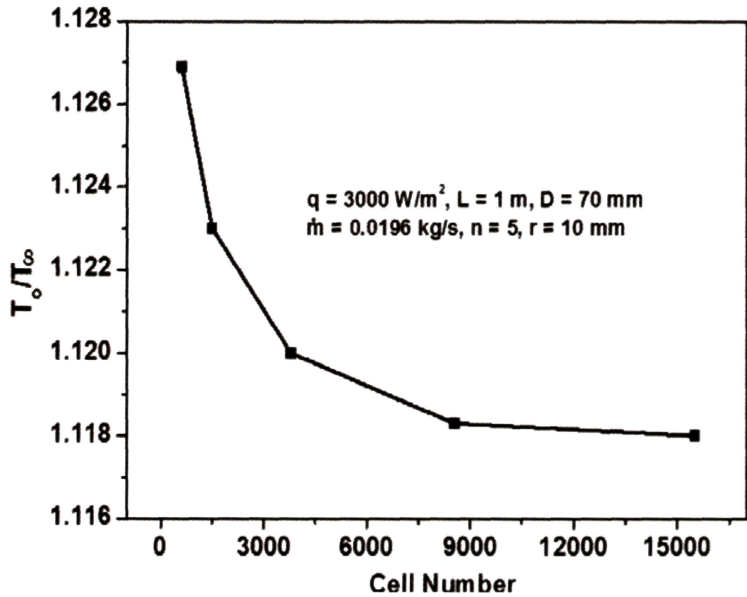


Figure 2: Outlet temperature as a function of cell number

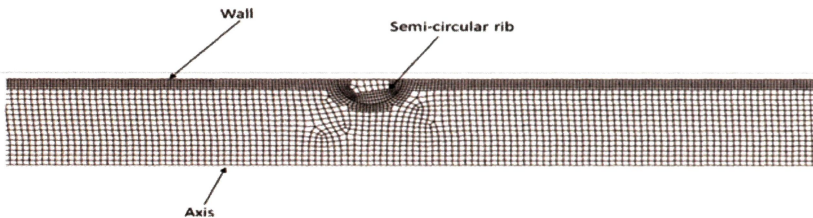


Figure 3: Cutaway of the meshes in and around the semi circular rib

### Rib height:

Wall temperature distribution as a function of rib height has been shown in the Figure 4. The rib is semi-circular in shape and attached to the wall at the initially was raised up to half of the tube where the rib is present and after that it suddenly dropped to a low temperature and in the downstream of rib the wall temperature is below compared to the plain tube. The rib creates an obstruction in flow, as a result of that the wall temperature sharply raised at the upstream of the rib and due to better turbulent mixing just after the rib the temperature is sharply dropped. It is also visualised from the plot when the height of the rib is increased the wall

### Rib number:

The effect of rib number on wall temperature has been shown in the Figure 6. The present numerical investigation the rib numbers was changed and its effect on wall temperature has been compared with plain tube. It can be seen from the plot the as the rib numbers are increased the average wall temperature decreases. This clearly indicates by providing more number of ribs the heat transfer to the air increases. This is also clearly visualised from Figure 7 where the Nusselt number is found to be more in case of higher numbers of ribs. Similar results were also found in Figure 8 when the shape of the ribs was changed from semi circular to rectangular. Here the average wall temperature is low for more number of ribs.

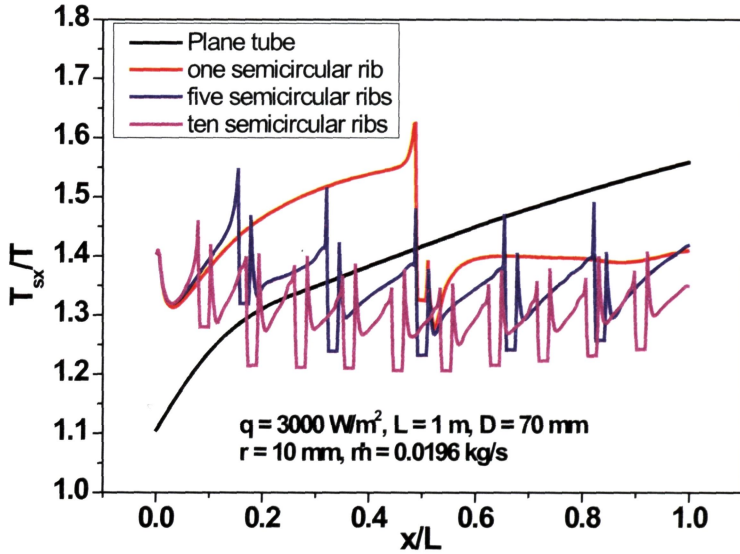


Figure 6: Wall temperatures as a function of number of semicircular ribs

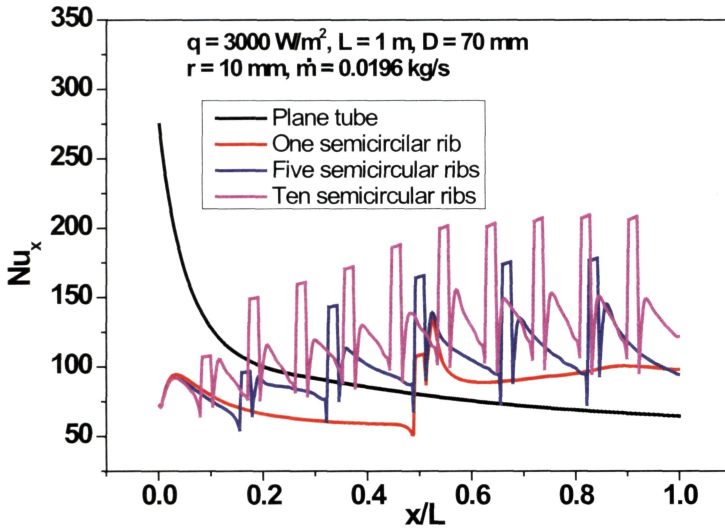


Figure 7: Nusselt number as a function of number of semicircular ribs

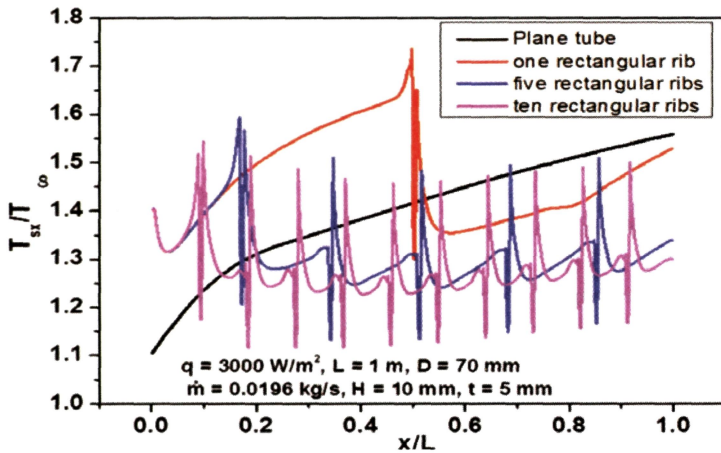


Figure 8: Wall temperatures as a function of number of rectangular ribs

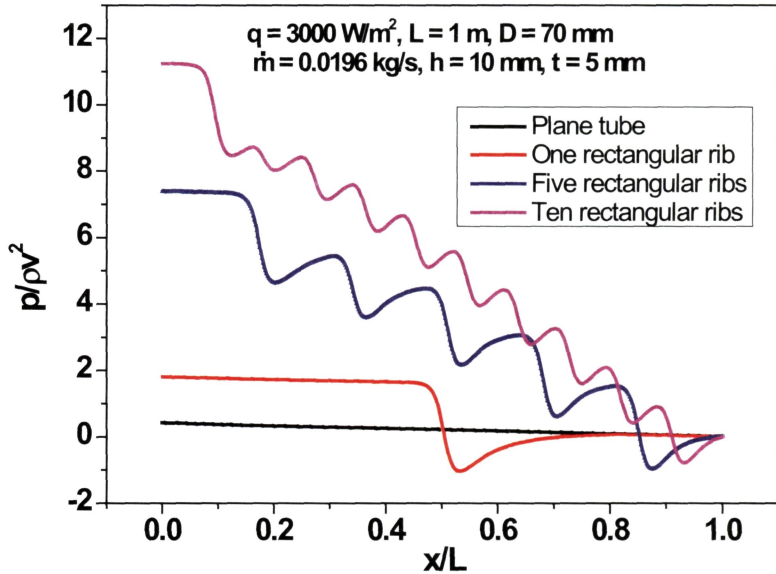


Figure 9: Centreline pressure as a function of number of rectangular ribs

So by providing more number of ribs, better turbulent mixing of the fluid would be achieved inside the tube which results in low wall temperature. Though the wall temperature is reduced considerably by providing more number of ribs, but the centreline pressure (Figure 9) is found to be high. This indicates more pumping power is required to deliver same amount of fluid.

### Rib inclination angle:

Rib inclination angle plays crucial role in heat transfer. The present numerical investigation is carried out for five number of rectangular ribs with inclination angle (a)  $45^\circ$ ,  $90^\circ$  and  $135^\circ$  with respect to the direction of flow as shown in the Figure 10. It can be seen from the plot the wall temperature is comparatively low for rib inclination angle  $45^\circ$  than other rib inclination angle which suggests the heat transfer to air is maximum at  $45^\circ$  inclination angle. The effect of increase in heat transfer is also clearly visible in Figure 11, where the Nusselt number is higher in case of  $45^\circ$  rib inclination angle. This means the convective heat transfer increases because of higher turbulent mixing when the ribs are inclined towards the upstream of flow. It is seen from Figure 12, the centreline pressure is low for  $45^\circ$  and  $135^\circ$  rib inclination angle compared to  $90^\circ$ . So power loss due to the friction is low and less pumping power is required to deliver the same quantity of fluid.



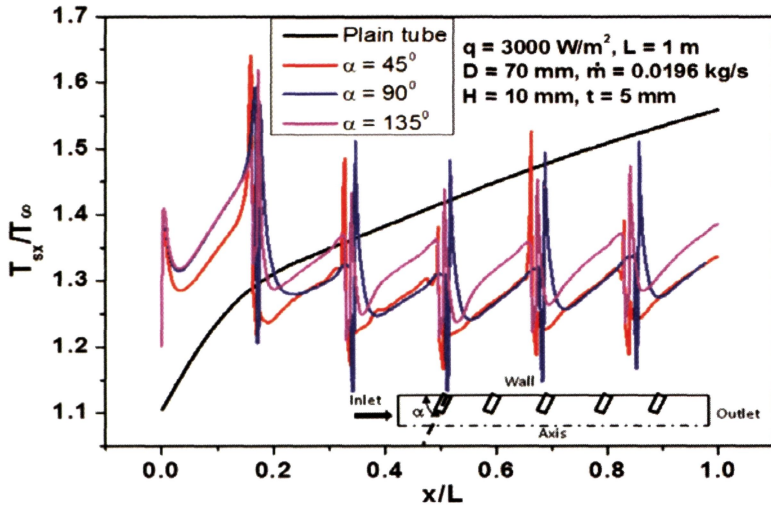


Figure 10: Tube wall temperatures as function rib inclination angle

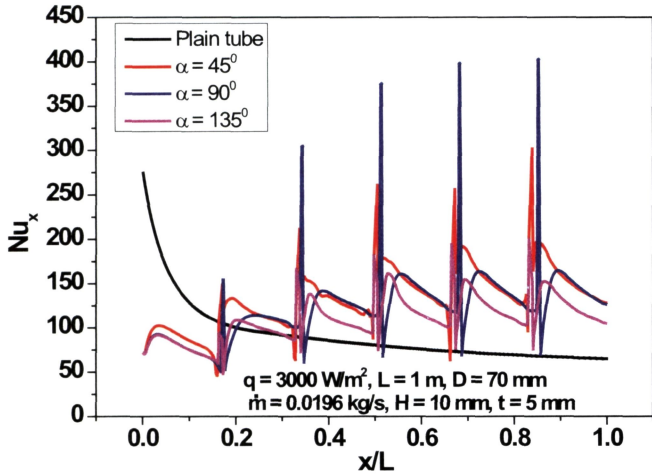


Figure 11: Tube wall temperatures as function rib inclination angle

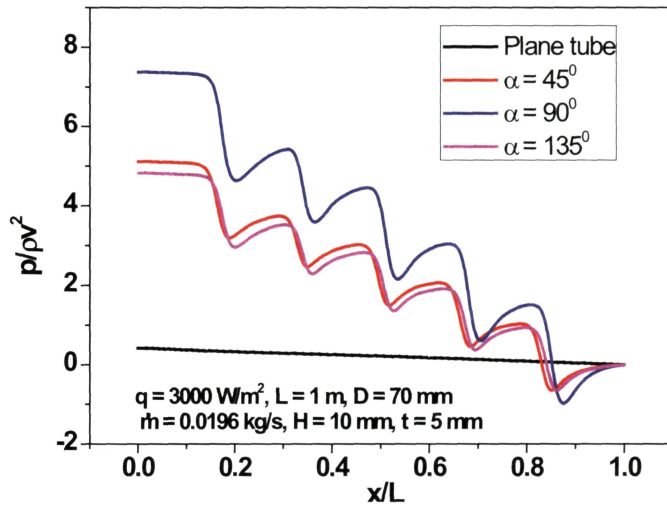


Figure 12: Tube centreline pressure as function rib inclination angle

### Rib shape:

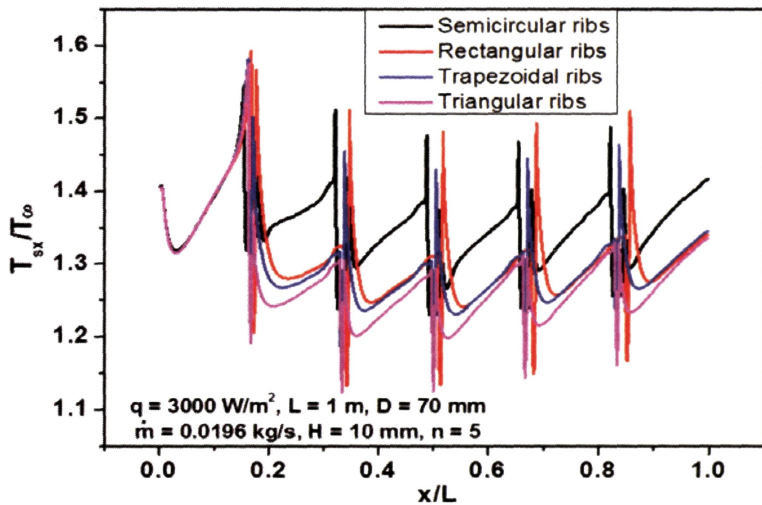


Figure 13: Tube wall temperatures as function shape of the ribs



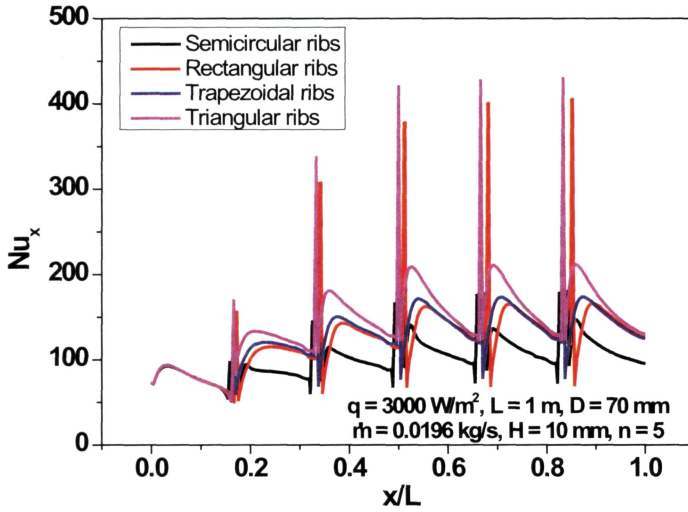


Figure 14: Nusselt number as function shape of the ribs

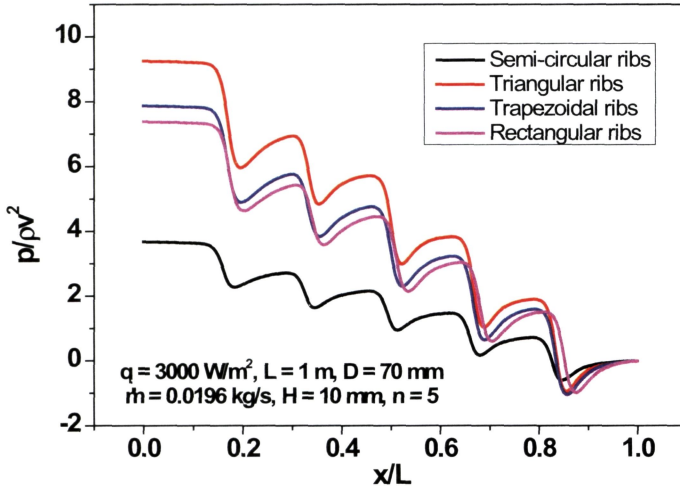


Figure 15: Tube Centreline pressure as function rib shape

The effect of rib shape on wall temperature has been shown in Figure 13. Five numbers of ribs of height 10 mm has been attached to the tube wall. The shape of the ribs in the present numerical investigation has been considered to be rectangular, semicircular, trapezoidal and triangular. It can be seen from the plot the wall temperature is low for triangular ribs, which implies the heat transfer more in

case of triangular ribs. This can be more prominently visualised in Figure 14, which shows the Nusselt number is high for triangular ribs. In triangular ribs the mixing of the fluid is better due to sharp edge and corner and heat transfer coefficient is high due to this turbulent mixing. But, it can be seen from Figure 15 the centreline pressure is high for triangular ribs because of sharp corner and edges.

## Conclusion

The wall temperature, tube outlet temperature and pressure drop for an internal ribbed tube have been computed by solving the conservation equations for mass, momentum and energy with two equation based k- $\epsilon$  turbulence model. The wall temperature decreases with the rib height and rib number. But the pressure drop is increased with the rib number. The heat transfer is found to be highest when the rib inclination angle is  $45^\circ$  with respect to the flow direction where the pressure drop is significantly less. The heat transfer rate was found to be highest in triangular ribs.

## References

- [1] R.L. Webb, E.R.G. Eckert and R.J. Goldstein, "Heat transfer and friction in tubes with repeated-rib roughness". *International journal of heat and mass transfer* 14(4), 601-617 (1971).
- [2] Lei Wang, Mirko Salewski and Bengt Sundén, "Turbulent flow in a ribbed channel: Flow structures in the vicinity of a rib." *Experimental thermal and Fluid Science* 34(2), 165-176 (2010).
- [3] S. Hossainpour and R. Hassanzadeh, "Numerical investigation of tub side heat transfer and pressure drop in helically corrugated tubes." *International Journal of Energy and Environmental Engineering* 2(2), 65-75 (2011).
- [4] Wei Peng, Jiang Pei-Xue, Wang Yang-Ping and Wei Bing-Yuan "Experimental and numerical investigation of convection heat transfer in channel with different types of ribs." *Applied Thermal Engineering* 3(14-15), 2702-2708 (2011).
- [5] Wen\_Tao Ji, Ding-Cai Zhang, He Ya-Ling and Tao Wen-Quan, "Prediction of fully developed turbulent heat transfer of internal helically ribbed tubes- An extension of Gnielinski equation". *International Journal of Heat and Mass Transfer*, 55, 1375-1384 (2012).
- [6] S. Anil Yadav and J.L. Bhagoria, "Heat Transfer and Fluid flow Analysis of an Artificial Roughened Solar Heater: A CFD Based Investigation". *Front Energy*, 8(2), 201-211 (2014).

- [7] K. Youngsiri, P. Eiamsa-ard, K. Wongcharee and S. Eiamsa-ard, "Augmented heat transfer in a turbulent channel flow with inclined detached-ribs." A case study in thermal engineering 3, 1-10 (2014).
- [8] K. Sivakumar, E.Natarajan and N. Kulasekharan, "Heat transfer and pressure drop comparison between smooth and different sized rib-roughened rectangular divergent ducts." *International Journal of Engineering and Technology (IJET)*, 6(1), 263-272 (2014).
- [9] K.R. Aharwal, C.B. Pawar and Alok Chaube, "Heat transfer and fluid flow analysis of artificially roughened ducts having rib and groove roughness." *Heat Mass Transfer*, 50(6), 835-847 (2014).
- [10] N. Shrirao Pankaj, M. Rahul Sherekar and V. Sachin Bhalerao, "Experimental analysis of turbulent flow heat transfer in a rectangular duct with and without continuous and discrete V-shaped internal ribs." *International Journal of Research in Advent Technology*, 2(1), 59-66 (2014).
- [11] S. K. Dash, "Heatline visualization in turbulent flow", *Int. J. Numerical Methods for Heat and Fluid Flow*, 6, 37-46 (1996)
- [12] P.K. Jha and S.K. Dash, "Effect of outlet positions and various turbulence models on mixing in a single and multi strand tundish", *Int. J. Num. Method for Heat and Fluid Flow*, 12, 560-584 (2002).
- [13] P.K. Jha and S.K. Dash, "Employment of different turbulence models to the design of optimum steel flows in a tundish", *Int. J. Numerical Methods for Heat and Fluid Flow*, 14, 953-979 (2004).
- [14] P.K Jha, R. Ranjan, S.S. Mondal and S.K. Dash, "Mixing in a tundish and a choice of turbulence model for its prediction", *Int. J. Numerical Methods for Heat and Fluid Flow*, 13, 964-996 (2003).
- [15] H.K. Versteeg and W. Malalasekera, "An Introduction to Computational Fluid Dynamics The Finite Volume Method", Addison Wesley Longman Limited, England (1996).
- [16] M.A. Islam and A.K. Mozumder, "Forced convection heat transfer performance of an internally finned tube", *Journal of Mechanical Engineering ME*, 40, 54-62 (2009).
- [17] B. Yu, J.H. Nie, Q.W. Wang and W.Q. Tao, "Experimental study on the pressure drop and heat transfer characteristics of tubes with internal wave-like longitudinal fins", *Heat and Mass Transfer*, 35, 65-73 (1999).